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ABSTRACT

This paper will present the two-phase condensation heat transfer coefficient and pressure drop characteristics of HCFC-22 and AZ-20[1] (an azeotropic mixture of 50 wt% HFC-32 and 50 wt% HFC-125). The experiments were conducted without oil in the refrigerant loop. The test section consists of a smooth, horizontal copper tube of 0.375 inches outside diameter (0.305 inches inner diameter) and 10 foot long. It is a counter flow heat exchanger with refrigerant flowing in the inner tube and water-glycol mixture flowing in the annulus. The average saturated condensing temperatures were 115°F and 125°F. The average inlet and exit qualities were 87% and 25% respectively. The mass flux was varied from 118 klb/ft^2-hr to 414 klb/ft^2-hr. A differential pressure transducer was used to measure the pressure drop across the test section. The results showed that at similar mass fluxes the condensation heat transfer coefficients for AZ-20 were slightly higher (about 2% to 6%) than those of R-22. However, the pressure drops for AZ-20 were significantly lower (about 25% to 45%) than those of R-22.

INTRODUCTION

HCFC-22 has long served as the working fluid in many air-conditioning systems. However, in accordance with the most recent revision of the Montreal Protocol at the Copenhagen conference (1992), HCFC-22 will be phased out early next century. As a result the search for replacing HCFC-22 has been intensified in recent years.

AZ-20, an azeotropic mixture of 50 wt% HFC-32 and 50 wt% HFC-125, has been considered as one of the primary replacements of HCFC-22 in air-conditioning system applications. Preliminary compressor calorimeter and system tests [2,3,4] have shown that AZ-20 offers a significant increase in energy efficiency over HCFC-22. Presently, there are limited AZ-20 heat transfer coefficient and pressure drop data available. The objective of this paper is to present oil-free two-phase flow experimental heat transfer and pressure drop data of HCFC-22 and AZ-20 in a smooth, horizontal copper tube.

TEST FACILITIES

The schematic diagram of the test rig is shown in Figure 1. It consists of two main loops: a refrigerant loop and a water-glycol loop. The test rig was designed to enable either the evaporation or condensation experiments.

Refrigerant Loop

The main components of the refrigerant loop consist of a receiver, variable speed refrigerant pump, mass flow meter, preheater (4.5 KW), evaporator (boiler 5.4 KW), condenser (test section), and after condenser.

The test section, as shown in Figure 2, is a counter flow heat exchanger with water-glycol mixture flowing in the annulus and refrigerant in the inner tube. The inner tube is a smooth, horizontal copper tube of 0.375 inches outside diameter (0.305 inches inner diameter) and 10 foot long. The outer tube is made of a clear PVC tube. The copper tube surface temperatures were measured directly by using thermocouples which were located on the top, side and bottom of the tube. Pressure drop across the test section was measured by using a differential pressure transducer. The tube length where the pressure drop was measured is 12 feet. The inlet and exit temperatures of the refrigerant were measured by using RTD sensors. The inlet pressure of the refrigerant was measured by using a pressure transducer. The refrigerant flow rate was controlled by adjusting the speed of the pump and was measured by using a mass flow meter. The test section is insulated by using a 1 1/2 inch thickness of armaflex tube insulation.

The electric power going into the preheater and evaporator were measured by watt transducers and adjusted by variacs. The after condenser is a counter flow heat exchanger with refrigerant flowing in the inner tube and water-glycol mixture in the annulus. It was used to condense the refrigerant leaving the test section.

Water-glycol Loop

The main components of the water-glycol loop consist of a variable speed pump, heater controlled by an SCR, 50 gallon tank, centrifugal pump, and R-502 chiller unit.

The inlet and exit temperatures of the water-glycol at the test section were measured by using RTD sensors. The flow rate was controlled by adjusting the speed of the pump and was measured by using a turbine flow meter. The temperature of the water-glycol at the test section was controlled by adjusting the power going to the heater via an SCR. Either the R-502 chiller unit or city water run through a heat exchanger was used as a heat sink for the test rig.

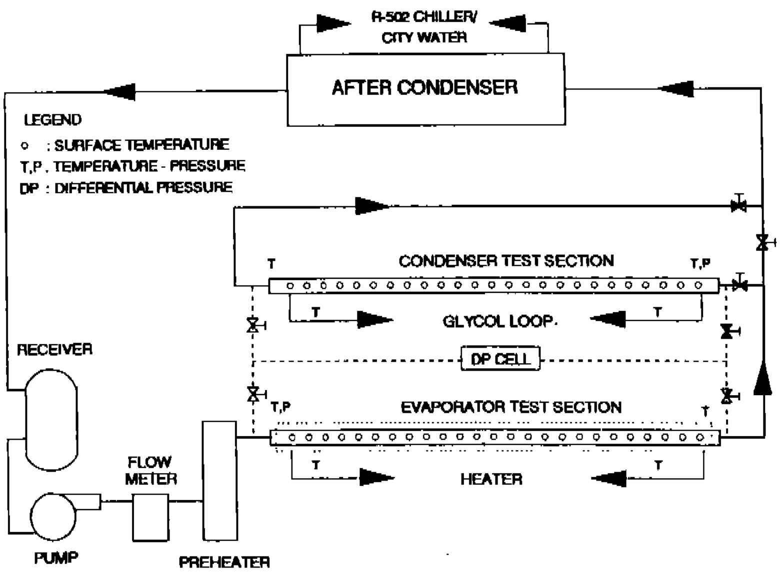

Figure 1. Schematic diagram of the heat transfer test rig

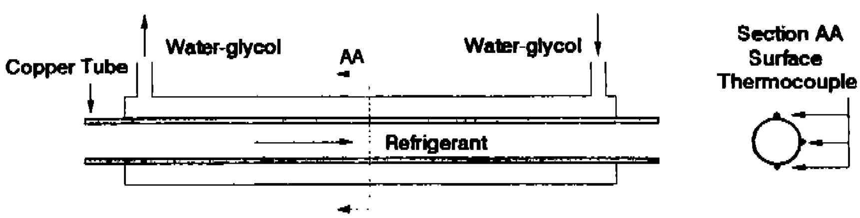

Figure 2. Schematic diagram of the condenser test section

Instrumentation Calibration

The corriolis mass flow meter was calibrated by running water at different flow rates and weighing the amount of water flowing in a given time period. The turbine flow meter was done in the same way except that the water-glycol mixture was used as the calibration fluid. The RTDs and thermocouples were calibrated in a refrigerated temperature bath. The pressure transducer was calibrated against a calibrated (reference) dial pressure gauge. The watt transducers were checked by measuring the voltage and current and comparing with the signal output.

The differential pressure transducer was checked by running a single-phase flow refrigerant and comparing its pressure drop values to those obtained by using the Gentry™ program[5] which was developed using a single phase-flow pressure drop equation. The deviations between the experimental data and the program were about 2% to 6%.

EXPERIMENTAL PROCEDURE

The thermodynamic properties of the refrigerant under test were incorporated in the data acquisition system. By knowing the inlet temperature and pressure of the refrigerant prior to the preheater and the power going into the preheater and evaporator, the inlet quality of the refrigerant in the test section can be calculated. The exit quality of the refrigerant can be determined by applying an energy balance on the water-glycol side. The system was allowed to come to steady state before any data was recorded. The steady state condition was reached when the refrigerant inlet pressure/temperature/quality, exit temperature/quality, mass flow rate and water-glycol mass flow rate, inlet and exit temperatures were not fluctuating. All channels in the data acquisition system were scanned five times and then averaged. The data collection took about 2 minutes for each run.

DATA REDUCTION

The heat transfer coefficient was computed by:

$$h = Q_g / [A_s * (T_{sat} - T_s)] \qquad Eq.2$$

where T_{sat} : the average refrigerant inlet and exit temperatures
T_s : the copper tube surface temperatures
Q_g : energy balance on the water-glycol side

$$Q_g = M_g * Cp_g * (T_{go} - T_{gi}) \qquad Eq.3$$

where Mg : water-glycol mass flow rate
Cp_g : water-glycol liquid specific heat
Tg : water-glycol temperature
subscripts o,i : outlet, inlet

The values of Q_g was checked by comparing the energy balance on the refrigerant side, Q_R, during single phase flow experiments. These two energy balances, Q_g and Q_R, agreed to within 4% for all runs.

TEST RESULTS

Experimental condensation heat transfer coefficient and pressure drop data are reported for R-22 and AZ-20. Table 1 shows the comparison of thermodynamic and transport properties of R-22 and AZ-20. The dP/dT, shown in Table 1., reflects the sensitivity of a given system's efficiency to pressure drop. The higher the value of dP/dT the lower the change of saturation temperature for a given pressure drop. Overall, AZ-20 shows better thermophysical properties than R-22. dP/dT of AZ-20 for the temperature range shown is about 55 % higher than R-22. At the condensing temperature range of 100 °F to 130 °F, the liquid thermal conductivity of AZ-20 is slightly lower than that of R-22; however, the vapor thermal conductivity of AZ-20 is higher. The liquid viscosity of AZ-20 is much lower (30% to 35%) than that of R-22 while its vapor viscosity is slightly higher than R-22. From these themophysical comparison, one may conclude that the condensation heat transfer and pressure drop characteristics of AZ-20 are equal or better than those of R-22.

Table 1. Comparison of thermophysical properties of R-22 and AZ-20

	% Difference of AZ-20 based on R-22			
Saturation Property	@ 20 F	@ 45 F	@ 100 F	@ 130 F
Pressure *	61.2	59.2	57.6	57.4
dP/dT *	55.9	55.8	56.9	56.9
Liquid thermal conductivity **	9.7	7.2	-0.3	-7.6
Vapor thermal conductivity **	13.2	12.5	10.2	9.1
Liquid viscosity **	-20.0	-23.3	-29.2	-35.5
Vapor viscosity **	1.5	2.1	5.5	10.8
Liquid density *	-8.3	-9.2	-12.8	-17.4
Vapor density *	42.3	42.7	49.8	60.3

* Reference Genie (6)

** Reference Refprop v 4.0 (7)

Heat Transfer Coefficient

As mentioned earlier, the average saturated condensing temperatures were 115 °F and 125 °F. The average inlet and exit qualities were 87% and 25% respectively. The mass flux was varied from 118 klb/ft^2-hr to 414 klb/ft^2-hr. For the local heat transfer coefficient, only the mass flux of 365 klb/ft^2 hr (3 lb/min) was studied. The local heat transfer coefficients obtained in the present study are compared with those generated by ACRC[8] as shown in Figure 3. Figure 3. shows a good agreement between the two data sources.

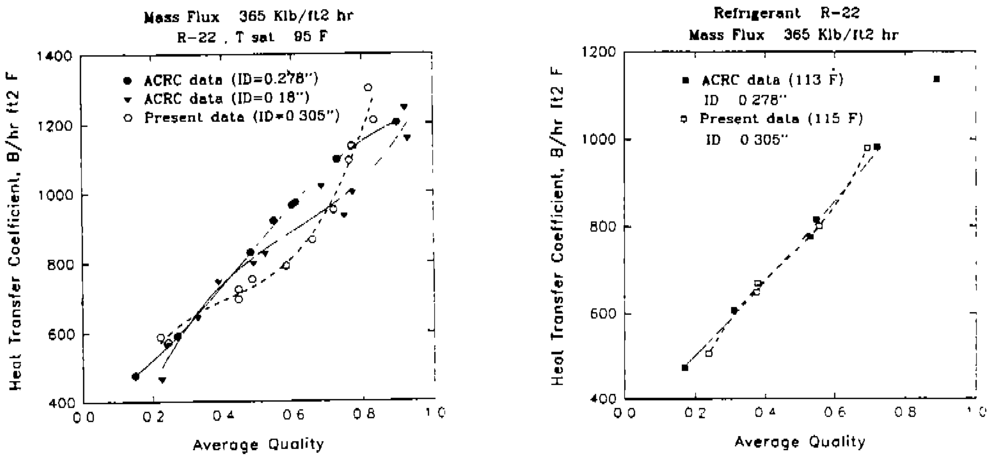

Figure 3. Comparison of local heat transfer coefficient

Figure 4 shows the local heat transfer coefficient data of R-22 and AZ-20 for the mass flow rate of 3 lb/min. For both refrigerants, the heat transfer coefficients increase with the quality and decrease with temperature. When the two refrigerants are compared to each other, AZ-20 shows slightly higher (about 3% to 6%) heat transfer coefficients at low refrigerant qualities for both condensing temperatures of 115 °F and 125 °F. At a lower quality, a refrigerant consists of more liquid than vapor. As shown in Table 1, the AZ-20 liquid thermal conductivity is slightly lower (0% to 8%) than that of R-22; however, its liquid viscosity is much lower (30% to 36%) than that of R-22. Therefore, this factor may result in higher heat transfer for AZ-20 at a lower refrigerant quality. Even though there is a consistent difference in local heat transfer at low refrigerant qualities between the two refrigerants for both 115 °F and 125 °F condensing temperatures, this difference may be well within the experimental uncertainties.

However, as the refrigerant quality increases the difference in the heat transfer coefficients between the two refrigerants diminishes.

Figure 5 shows the overall heat transfer coefficient data of the two refrigerants for different refrigerant mass flow rates and condensing temperatures. For both refrigerants, the heat transfer coefficients increase with mass flow rate and decrease with temperature. As shown in Figure 5, AZ-20 shows slightly higher (about 2% to 6%) heat transfer coefficients than R-22. Again, this difference in the overall heat transfer coefficient may be well within the experimental uncertainties.

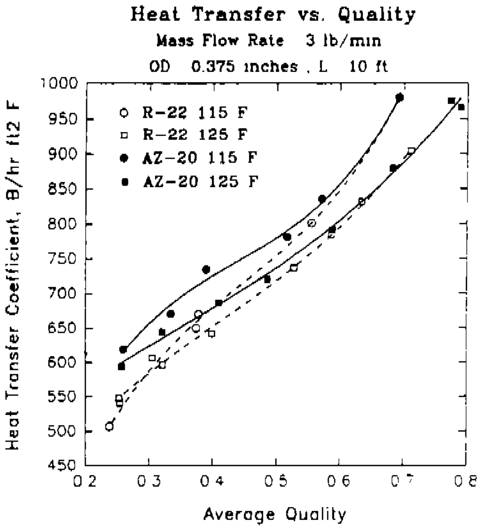

Figure 4. Local heat transfer coefficient

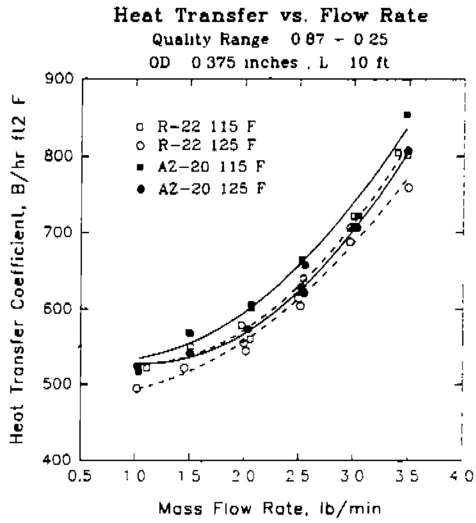

Figure 5. Overall heat transfer coefficient

Pressure Drop

As described earlier, a differential pressure transducer was used to measure the pressure drop across the test section. The actual tube length where the pressure drop was measured was 12 ft. Figure 6 shows the pressure drop versus quality during condensation for R-22 and AZ-20 while Figure 7 shows the pressure drop versus refrigerant mass flow rate for both refrigerants at different condensing temperatures.

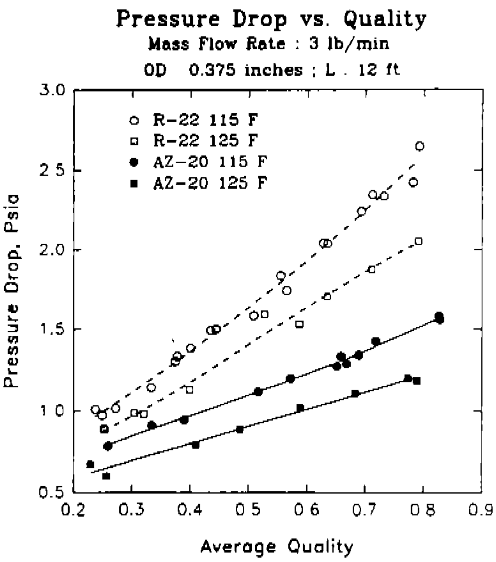

Figure 6. Pressure drop vs. quality

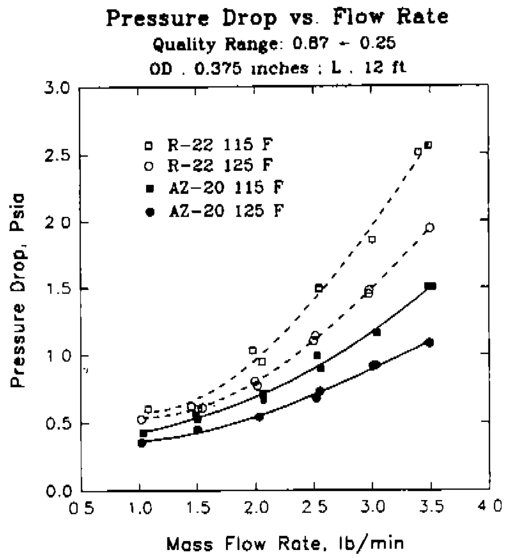

Figure 7. Pressure drop vs. mass flow rate

As shown in Figure 6, for both refrigerants the pressure drops increase with quality and decrease with temperature. At both condensing temperatures of 115°F and 125°F, AZ-20 results in significantly lower pressure drops (about 20% to 40%) than R-22. The differences in pressure drop between the two refrigerants become more significant at higher quality. At a higher quality, a refrigerant contains more vapor than liquid. As shown in Table 1, vapor density of AZ-20 is much larger (about 40% to 60 %) than that of R-22. The larger the vapor density the smaller the pressure drop a refrigerant will experience.

Figure 7 show the pressure drop versus mass flow rate. For both refrigerants the pressure drops increase with refrigerant mass flow rate and decrease with temperature. AZ-20 results in significantly lower (about 25% to 45%) pressure drops than R-22. The differences in pressure drop become more significant as the refrigerant mass flow rate increases.

CONCLUSIONS

Condensation heat transfer coefficient and pressure drop were experimentally measured for both refrigerant R-22 and AZ-20. The heat transfer coefficients of AZ-20 were slightly higher than those of R-22. The differences may be within experimental uncertainties. On the other hand, the pressure drops of AZ-20 were significantly lower than those of R-22. The lower pressure drop characteristics and bigger dP/dT values of AZ-20 can be very beneficial in improving the efficiency of an air-conditioning/heat pump system. A condenser tube size can be reduced significantly to increase the refrigerant mass flux. As a result, a higher refrigerant mass flux, as shown in Figure 5, will enhance the heat transfer coefficient thus increase the system energy efficiency.

ACKNOWLEDGMENT

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